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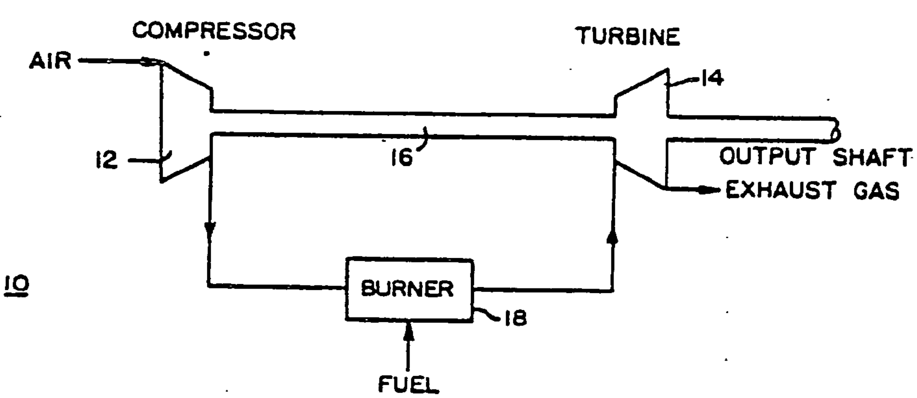

(57) Abstract

A constant pressure Brayton cycle engine (40) including a combustion chamber (46) and a positive displacement compressor (42) connected by a shaft (41) to an output wheel (44). Gas is fed to the compressor (42), then to the combustion chamber (46) and then to the output wheel (44). The engine (40) can be either open cycle or closed cycle. The compressor (42) can be of any suitable type such as a sliding vane, a Roots type blower (sometimes called a screw or gear compressor), a regenerative blower, or a liquid seal (Nash) blower. The output wheel (44) can be any of these types or also a dynamic (axial or centrifugal) turbine wheel.

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ENGINE

CROSS-REFERENCE TO RELATED APPLICATIONS

This application is a continuation-in-part of applicant's copending application serial No. 879,969, filed February 21, 1978 and entitled "Gas Turbine System"; and is also a continuation-in-part of applicant's copending application serial No. 890,465, filed March 27, 1978 and entitled "Gas Turbine System". Each of these two parent applications in their entirety is hereby incorporated by reference in this application.

TECHNICAL FIELD

This invention relates to engines or prime movers and in a preferred embodiment to a constant pressure Brayton cycle engine using a positive displacement compressor.

BACKGROUND OF THE PRIOR ART

All previous methods of adapting the Brayton (or Joule or constant pressure) continuous cycle for prime mover service have centered around the classical gas turbine concept. A gas turbine includes three devices: A dynamic (non-positive displacement) compressor of the axial or centrifugal type; a fuel combustor of the direct of indirect type; an output turbine (power wheel) of the impulse or reaction type. The gas turbine approach has many advantages at high power levels (central station power generation, aircraft engines, etc.), but suffers serious economic handicaps at the lower horsepower levels (up to about 300 h.p.) required by automobile service.

The compressor and turbine share a common shaft in the classical gas turbine engine. Thus, the turbine (in a single shaft engine) must drive its own compressor in additional to driving the load. The reason for this is that only a gas turbine is capable of the ultra-high r.p.m. required by the previously used, high speed, dynamic compressors. A simple single shaft, classical engine is shown in Fig. 1

and comprises a compressor 12, a turbine 14, a shaft 16 connecting the compressor and turbine and a combustion chamber 18. Fig. 2 shows two cases of different compressor-to-output power ratios (Pc/Po). Fig. 2A shows a compressor 20 and a turbine 22 connected together by a shaft 24. The arrangement shown in Fig. 2A has a compressor-to-output power ratio of 0.5. Thus, a 100 h.p. output requires a 0.5 x 100 = 50 h.p. compressor. It follows that the power turbine for this case must supply 50 + 100 = 150 h.p. Fig. 2B shows a compressor 26 and a turbine 28 connected together by a shaft 33. In Fig. 2B the compressor-to-output power ratio is 2.0. The 100 h.p. output power for this case requires a 200 h.p. compressor and a 100 + 200 = 300 h.p. turbine. Clearly the overall equipment size, for a given output shaft power, depends upon the required compressor size. It can be shown (see "Aircraft Gas Turbines" C. W. Smith John Wiley - page 43 or "Propulsion Systems" A. N. Hosny, University of South Carolina Press - page 51) that the compressor to output power ratio can be simply stated as:

$$1. \quad \frac{Pc}{Po} = \frac{1}{\eta_T \, \eta_C \, \frac{1}{x} \left(\frac{T3}{T1}\right) - 1}$$

where:

Pc/Po = Compressor To Output Horsepower Ratio

T_3/T_1 = Combustion To Inlet Temperature Ratio (^oF, Absolute)

η_T = Thermodynamic Efficiency of Turbine

η_C = Thermodynamic Efficiency of Compressor

$x = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$

P_2/P_1 = Compressor Outlet To Inlet Pressure Ratio

γ = Ratio of Heat Capacity At Constant Pressure To That At Constant Volume

For a fixed ambient temperature (T_1) equation 1 states that the compressor-to-output power ratio depends only upon the pressure ratio adopted and the combustion temperature, since the other parameters, η_T, η_c, and γ are essentially

fixed. Equation 1 is plotted in Fig. 3 and provides a good overall picture of the classical problem. Anything above a compressor-to-output power ratio of 1.0 means a very large compressor indeed. The parameters from Fig. 3 can be transformed to practical values in order to illustrate the classic problem involved and the solution provided by this invention. Consider the case of 50 h.p. net output.

The data from Fig. 3 can now be translated to absolute compressor ratings and this is plotted in Fig. 4 Typical materials in today's technology allow for a temperature ratio (T_3/T_1) of about 3. Thus, Fig. 4 shows a marked sensitivity to pressure ratio. Absolute efficiency considerations dictate a pressure ratio in the range of 4:1 to 6:1. If a temperature ratio of 6 could be tolerated, Fig. 4 shows an insensitivity of compressor requirements to pressure ratio. If we assume an ambient temperature of 70°F, then a clearer picture of the problem emerges and is shown in Fig. 5. The steepness of the curve in the vicinity of 1000°F is the cause of the major problems as will become clear from the next curve.

A careful review of commercially available compressors of different types allows the compressor weight vs. horsepower curves of Fig. 6 to be drawn. The high r.p.m. associated with classical gas turbines limits severely the tolerable turbine inlet temperature. The relatively cheap materials that must be used for automobile service sets a limit of 1000-1500°F. Fig. 5 dictates a compressor in the 50-100 h.p. class to supply a net output of 50 h.p. Consider now Fig. 6: A piston compressor is completely out of the question since it would weigh between 1500 and 2000 pounds. The Roots type blower and sliding vane type compressors are better but also are in the impractical range of 200-600 pounds to supply 50 h.p. of shaft output power. Only the non-positive displacement compressor has a reasonable weight for the classical approach. It is this compressor weight factor alone - not cost nor efficiency but simply compressor weight that forces classical gas

turbines to use the non-positive displacement compressor to supply the massive amounts of compressed air that are required.

The penalty paid for achieving the high compressor h.p. to weight ratio for the classic gas turbine is severe indeed:

1. The ultra-high r.p.m. is many times greater than the automobile r.p.m.

2. The ultra-high r.p.m. of the hot turbine gives rise to excessive tensile stress due to centrifugal forces. This limits both usable materials and allowable inlet temperatures.

3. The compressor mass flow and output pressure vary with r.p.m. The fact that pressure varies means part-load efficiency is poor.

It is important to note that the output turbine requirement does not dictate a high r.p.m., per se; the high r.p.m. is required strictly to satisfy the compressor needs. In fact, there is no other method (electric motor drive, for example) of achieving the high r.p.m. required by the compressor.

This then is the state of affairs for gas turbines in automobile service. Fuel economy dictates both a high inlet temperature and a relatively high pressure ratio (about 4:1). Material limitations prevent high combustion temperatures, which in turn results in both low efficiency and high compressor requirements (Fig. 5), thus practically eliminating all but high speed, turbo-compressors (Fig. 6) on a weight basis alone. It is important to note that jet aircraft, central station power, and other non-auto applications can afford to use more expensive materials and devices and hence can utilize somewhat higher combustion temperatures. A jet aircraft turbine may cost $50,000 to

build, but an automobile gas turbine must be limited to the $100-$200 class. As a result, gas turbines have found widespread service outside the automobile field only.

The main thrust of gas turbine technology today is towards better and cheaper high temperature materials and assemblies, super-alloys, exotic cooling methods, ceramic turbine buckets, cermet technology, plasma coatings, etc.

SUMMARY OF THE INVENTION

The present invention is a Brayton cycle engine method and apparatus using a positive displacement compressor. The Brayton cycle engine of the present invention includes a positive displacement compressor, an output wheel connected by a shaft to the compressor, a combustion chamber, and means for feeding a gas to the compressor, means for feeding compressed gas from the compressor to the combustion chamber, means for burning fuel in the combustion chamber, means for feeding hot gas from the combustion chamber to the output wheel, and means for feeding exhaust gas from the output wheel. The compressor can be any suitable positive displacement compressor such as a sliding vane compressor, a Roots type blower (sometimes called a screw or gear compressor), a regenerative blower, or a liquid seal (Nash) type blower. The output wheel can be any one of the above mentioned devices or in addition can be a dynamic turbine wheel.

This removes the classic problems associated with high speed, non-positive displacement compressors: high cost and complexity, high stress on hot turbine buckets, low efficiency at part-load, etc. Use of a low speed, positive displacement compressor is made possible by the use of high combustion temperatures as shown in Fig. 5. These high combustion temperatures are

basically made possible by either one of two methods:

1. Use of an intermittent burn-cool operating cycle as described in copending application serial No. 879,969; and
2. Use of a bootstrap argument. This results from the basic understanding provided by Figs. 1 through 6. The adoption of a lower r.p.m. means a lower tensile stress on the hot turbine bucket. The reduced stress permits an increased temperature since high temperature creep is a main limitation. The increased temperature results in reduced compressor requirements (Fig. 5) which in turn makes positive displacement compressors (low r.p.m.) feasible. The net result is a cheaper, more efficient and flexible implementation of the Brayton cycle.

Of course, as new high temperature materials, devices, and techniques became available they can be incorporated into the present inventions. However, it is to be clearly understood that the present invention can be practiced with today's materials, devices, and technologies.

It is desirable to expand Fig. 6 to the 0-50 h.p. range of interest to automobile service. This results in Fig. 7. However, since Roots blowers, sliding vane compressors, helical screw compressors, etc. have not been engineered with the present problem in mind, it is felt that improvements can be readily made to transform the data in Fig. 7 into the curves of Fig. 8. The basic idea to be conveyed is that virtually any type of positive displacement compressor can be used to implement the present invention.

It is an object of the present invention to overcome the disadvantages of the Brayton cycle gas turbine

by providing a non-gas turbine implementation of the constant pressure Brayton cycle. It is another object of the invention to provide a Brayton cycle engine having a positive displacement compressor.

BRIEF DESCRIPTION OF THE DRAWINGS

The present invention will be more fully understood by reference to the following detailed description thereof, when read in conjunction with the attached drawings, wherein like reference numerals refer to like elements and wherein;

Fig. 1 is a partly diagramatic, partly schematic view of a prior art Brayton cycle gas turbine;

Figs. 2A and 2B are diagramatic views of a compressor-turbine-shaft combination;

Fig. 3 is a graph of the ratio of compressor power to output shaft power vs. pressure ratio.

Fig. 4 is a graph of required compressor power (h.p.) vs. pressure ratio;

Fig. 5 is a graph of required compressor power (h.p.) vs. combustion temperature (°F);

Fig. 6 is a graph of compressor weight (lbs.) which is compressor power (h.p.);

Fig. 7 is a graph of compressor weight (lbs.) vs. compressor power (h.p.);

Fig. 8 is a graph of compressor weight (lbs.) vs. compressor power (h.p.);

Fig. 9 is a partly diagramatic, partly schematic view of an engine according to one embodiment of the present invention including a compressor, an output power wheel and a combustion chamber;

Fig. 10 is a partly diagramatic, partly schematic, view of an engine according to another embodiment of the present invention including a compressor, an output power wheel and an indirect combustion chamber for use in a closed cycle operation;

Fig. 11 is a graph showing desirability vs. horsepower, speed, or torque;

Fig. 12 is a graph showing tensile strength vs. temperature for selected graphites;

Fig. 13 is a graph showing vaporization rate vs. temperature for graphite;

Fig. 14 is a graph showing pressure and flow rate vs. r.p.m. for a dynamic turbo-compressor; and

Fig. 15 is a graph showing pressure and flow rate vs. r.p.m. for a positive displacement compressor.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

With reference to the drawings, Fig. 9 shows a substantially constant pressure Brayton cycle engine 40 according to the present invention comprising a positive displacement compressor 42 and an output power wheel 44 connected to the compressor by a shaft 41 and also connected to an output shaft 48. The engine 40 includes a combustion chamber 46 to which fuel is fed by a fuel line 47. Air is fed into the compressor 42 through an air inlet 43 and compressed air is fed from the compressor 42 to the combustion chamber 46 through an air line 49. Compressed, hot gas is fed from the combustion chamber 46 through a continuation of the gas line 49 to the output power wheel 44 from which the exhaust gas is fed to ambient through an exhaust line 45. The engine 40 is an open cycle engine as will be clearly understood by those skilled in the art.

Fig. 10 shows a closed cycle engine 50 according to another embodiment of the present invention. The engine 50 includes a positive displacement compressor 52 and an output power wheel 54 connected to the compressor by a shaft 56. The output power wheel 54 is also connected to an output shaft 60. The engine 50 also includes an indirect combustion chamber 58 into which air and fuel are fed by lines 68 and 70 respectively, and from which

the exhaust gas is fed to ambient through an exhaust line 72. Compressed gas from the compressor 54 is fed by a line 62 from the compressor to a heat exchanger 74 (such as a coil) in the indirect combustion chamber 58 and from there to the output power wheel 54. The exhaust from the output wheel 54 is fed by a line 64 back to the compressor 52. The exhaust gas from the output power wheel 54 is heat exchanged by a heat exchanger 66 with the gas fed from the compressor 52 to the combustion chamber 58. The various advantages of the closed cycle are described in the parent applications incorporated herein by reference above. It is to be noted that in the open cycle embodiment of Fig. 9, the hot gas fed to the output power wheel is the products of combustion of the burning fuel in air. In the closed cycle embodiment of Fig. 10, on the other hand, the hot gas fed to the output power wheel is whatever is chosen for the working gas, such as nitrogen, neon, carbon dioxide, etc.

According to the present invention, the engines 40 in Fig. 9 and 50 in Fig. 10 can use for the compressor any positive displacement compressor such as, for example: (1) a sliding vane compressor, (2) a Roots type blower (sometimes called a screw or gear compressor), (3) a regenerative blower, or (4) a liquid seal (Nash) type blower.

The output power wheel in each embodiment can also use any one of the above listed types of devices and in addition can use a dynamic turbine wheel. Thus, 20 different combinations are preferred and each has its own special application. For example:

1. Rotary vane/Rotary vane - is preferred for small car automotive service;
2. Rotary vane/Roots blower - is preferred for truck service;

3. Roots blower/Roots blower - is preferred for low speed, high torque service (farm tractor, etc.)

In the above, the first name refers to the compressor and the second to the output power wheel or motor. Thus, a rotary vane/roots blower configuration means a rotary vane compressor and a roots blower run backwards as a motor. It should be noted that compressors run backwards act as efficient motors, see, for example, "Pneumatics and Hydraulics" by H. L. Steward (Audel and Co.) illustrate this point. The Gast Manufacturing offers sliding vane motors for sale in the 1-10 h.p. class (Models 16AM-FCC-1).

These configurations result in a desirability curve vs. horsepower and/or speed and/or torque as depicted in Fig. 11.

One preferred specific embodiment is to use a sliding vane motor (output power wheel) with the vanes being made of graphite. This offers several advantages. Graphite is a good lubricant in itself and thus the wear on the vanes is minimized. Second, graphite is known to increase in tensile strength as temperature is increased to about 4500°F as shown in Fig. 12. Thus, this feature allows combustion temperatures up to about 4500°F to be used in the expansion output wheel. Fig. 12 is a graph of ultimate tensile strength vs. temperature for selected graphites. All specimens were tested in the direction of major anisotropy. (1) Petroleum coke base, fine grain, extruded, d = 1.67; (2) lampblack base, molded, d = 1.50; (3) petroleum coke base, medium grain, extruded, d = 1.55; (4) petroleum coke base, fine grain, molded, d = 1.75.

Graphite is known to have a relatively high vapor pressure at high temperatures. This will result in a vaporization rate. Assuming 100,000 miles of service life, and 50 miles/hour average speed, then 7×10^6 sec.

of service life is required. If 10% of the graphite surface is allowed to vaporize during this period, then a vaporization rate of about 10^{-8} gram/cm^2 sec. is tolerable. From Fig. 13 (which is the free vaporization rate of graphite) it is seen that an average temperature of about 3600°F (2200°K) is tolerable. While this temperature greatly exceeds present day gas turbine capabilities, it can be extended further by coating the graphite vanes with a suitable high temperature metal, alloy, or ceramic. In this case, vaporization will occur (if at all), only at the tiny exposed area where the vane rubs the housing.

While the above discussion has been with respect to using graphite for the sliding vanes in the motor or output power wheel of the present invention, it is also desirable to use graphite for the sliding vanes in a sliding vane type compressor. Graphite has the advantages that it wears uniformly and conforms by the wear process to the desired shape and that it also provides self lubrication, is relatively inexpensive and is easy to machine, etc.

Another method of causing a major reduction in the compressor size and weight is to utilize the closed cycle in Fig. 10. This technique does not change the compressor horsepower requirement. This fact is evident from equation 1. The compressor-to-output power ratio depends only upon the pressure ratio and not the absolute gas pressure. However, at higher total pressures made possible by the closed gas system, the compressor "swept out volume" becomes less due to the higher gas density. As a result, the key item (the compressor weight) is reduced. This allows for an extension of this invention to higher power levels. Thus, for example, the vane/vane desirability curve can now take the dashed position shown in Fig. 12.

It should be noted that a turbine does not require the same high tip speed (high r.p.m.) as does the turbo-compressor. Thus, combinations like vane/turbine and roots blower/turbine are useful configurations. While the main thrust of this invention is the automobile, it is to be understood that the engine of this invention is also applicable to trucks, locomotives, central station power, remote station power generation, emergency and stand-by power generators, aircraft, etc.

The present invention can be used with multiple shaft arrangements, multiple stage compressors, multiple stage expansion stages (motor output), interstage cooling, re-heat, and various techniques of heat exchange. The intermittent burn-cool operation cycle of parent application serial No. 879,969, the throttle control invention of parent application serial No. 890,465, the various flywheels and compressed gas surge tanks, etc. taught in said parent applications are also applicable in this invention.

One major advantage of the present invention over the classical gas turbine for automotive service is in the matching of r.p.m. The wheels of an automobile rotate at a maximum of about 1500 r.p.m. Thus, complicated speed reducers are required to match a gas turbine, at 50,000 r.p.m., to an automobile. The r.p.m. of the present invention can be selected to closely match that of the automobile by proper selection of compressor and power wheel diameters.

Another major advantage of this invention over the prior art is the efficiency at part-load. In fact, the present teachings maintain efficiency essentially down to 0 r.p.m. This statement can best be visualized with the aid of Figs. 14 and 15. The classical turbo-compressor relies upon high impeller tip speed to provide both mass flow and pressure. Thus, a reduction in r.p.m. causes both mass flow rate and pressure to be reduced as shown in

Figs. 14. The flow rate reduction at low r.p.m. is acceptable since low r.p.m. (part-load) requires only a portion of the air mass flow. However, efficiency considerations insist upon a high gas pressure, in fact, optimal conditions require a narrow band of gas, pressures (pressure ratio in 4-6 range). Thus, the low r.p.m. drops the pressure ratio and introduces the well-known part-load inefficiency. This is not true for the positive displacement compressor of the present invention as shown in Fig. 15. The compressor outlet pressure does not depend upon speed but only upon the geometry of the compressor apparatus. The flow rate is a direct linear function of the r.p.m. This is the ideal state of affairs for part-load. As more fuel is pumped to the combustor, the power output wheel (sliding vane, turbine, more output power and higher r.p.m. The higher r.p.m. provides more compressed gas, as it should, at the same pressure and thus the efficiency, and combustion temperature conditions do not change.

The following chart may be found useful:

TYPICAL COMMERCIAL UNITS

Mfg.	Type	Horsepower		Weight (pounds)	Model No.
		Room Temp.	2500° F*		
Gast	Rotary vane	4.0	20.0	21.0	6AM-FRV-23A
Gast	Rotary vane	10.0	50.0	80.0	16AM-FRV-13
Roots (Dresler)	Roots blower	40.0	200.0	600.0	RAI-88

*Estimated, as motor

The invention has been described in detail with particular reference to the preferred embodiments thereof, but it will be understood that variations and modifications can be effected within the spirit and scope of the invention as described hereinabove and as defined in the appended claims.

What is claimed is:

1. A Brayton cycle engine comprising:

(a) a positive displacement compressor having a gas inlet and a gas outlet;

(b) a combustion chamber including an inlet and an outlet and means for burning a fuel therein;

(c) an output wheel having a gas inlet and a gas outlet and being connected to said compressor by a shaft;

(d) means for feeding a gas to said compressor inlet, means for feeding compressed gas from said compressor outlet to said combustion chamber inlet; means for feeding hot compressed gas from said combustion chamber outlet to said output wheel inlet, and means for feeding exhaust gas out of said output wheel outlet; and

(e) means for operating said engine in a substantially constant pressure brayton cycle.

2. The apparatus according to claim 1 wherein said operating means comprises means for operating said engine in an open cycle and wherein said exhaust gas feeding means, feeds said exhaust gas to ambient.

3. The apparatus according to claim 1 wherein said operating means comprises means for operating said engine in a closed cycle and wherein said exhaust gas feeding means includes means for feeding said exhaust gas back to said compressor inlet, and wherein said combustion chamber is an indirect combustion chamber.

4. The apparatus according to claim 3 including means for heat exchanging the exhaust gas with the gas fed from said compressor to said combustion chamber.

5. The apparatus according to claim 1 wherein said compressor is a sliding vane compressor.

6. The apparatus according to claim 1 wherein said compressor is a gear type compressor.

7. The apparatus according to claim 1 wherein said compressor is a regenerative blower.

8. The apparatus according to claim 1 wherein said compressor is a liquid seal blower.

9. The apparatus according to any one of claims 5, 6 , 7 or 8 wherein said output wheel is a sliding vane output wheel.

10. The apparatus according to any one of claims 5, 6, 7, or 8 wherein said output wheel is a gear type output wheel.

11. The apparatus according to any one of these claims 5, 6, 7, or 8 wherein said output wheel is a regenerative blower output wheel.

12. The apparatus according to any one of claims 5, 6, 7, or 8 wherein said output wheel is a liquid seal blower output wheel.

13. The apparatus according to any one of claims 5, 6, 7, or 8 wherein said output wheel is a dynamic turbine.

14. The apparatus according to any one of claims 5, 6, 7, or 8 wherein said operating means comprises means for operating said engine in a closed cycle and wherein said exhaust gas feeding means including means for feeding said exhaust gas back to said compressor inlet, and wherein said combustion chamber is an indirect combustion chamber.

15. The apparatus according to claim 1 wherein the sliding vanes of said compressor are made of graphite.

16. The apparatus according to claim 9 wherein the sliding vanes of said output wheel are graphite.

17. The apparatus according to claim 1 wherein said output wheel includes a rotating member and wherein said operating means includes means for generating a hot-cool operating cycle comprising a hot phase continuously alternating with a cool phase, the gas fed to said output wheel during said hot phase being hot and the gas fed to said output wheel during said cool phase being cooler, said hot phase having a time period less than the time required for said rotating member to soak to substantially the temperature of the hot gas fed to said output wheel during the hot phase.

18. The apparatus according to claim 17 including means for maintaining the speed of said shaft during said cool phase substantially the same as that of said shaft during said hot phase, for providing uniform and continuous output.

19. The apparatus according to claim 17 including means for varying the length of at least one of said hot and cool phases in response to throttle changes.

20. The apparatus according to claim 1 including means for maintaining the temperature of the gas flowing through said output wheel gas inlet higher than the temperature of the gas at any other location in said engine.

21. The apparatus according to claim 1 including means for maintaining the burn termperature in said combustion chamber substantially constant even as the rate fuel is fed to said combustion chamber varies.

22. A method for operating a brayton cycle engine having a combustion chamber and a positive displacement compressor connected by a shaft to an output wheel, said method comprising the steps of feeding a gas to said compressor, feeding compressed gas from said compressor to said combustion chamber, burning fuel in said combustion chamber to heat the gas fed thereto from said compressor,

feeding hot, compressed gas from said combustion chamber to said output wheel, feeding exhaust gas away from said output wheel, and operating said engine in a substantially constant pressure, Brayton cycle.

23. The method according to claim 22 including operating said engine in an open cycle, including the step of feeding said output gas to ambient.

24. The method according to claim 22 wherein said engine is a closed cycle engine including a closed conduit for continuously circulating a working gas in a closed cycle from said compressor to said output wheel and then back again to said compressor, and wherein said combustion chamber is an indirect combustion chamber, and including the step of indirectly heating said working gas in said combustion chamber.

25. The method according to claim 24 including the step of heat exchanging the exhaust gas from said output wheel with the gas fed from said compressor to said combustion chamber.

26. The method according to claim 22 including the step of compressing said gas in a sliding vane compressor.

27. The method according to claim 22 including compressing said gas in a gear type compressor.

28. The method according to claim 22 including compressing said gas in a regenerative blower.

29. The method according to claim 22 including the step of compressing said gas in a liquid seal blower.

30. The method according to any one of claims 26, 27, 28 or 29 including the step of feeding the gas from said combustion chamber to a sliding vane output wheel.

31. The method according to any one of claims 26, 27, 28, or 29 including the step of feeding the gas from said combustion chamber to a gear type output wheel.

32. The method according to any one of claims 26, 27, 28 or 29 including the step of feeding the gas from said combustion chamber to a regenerative blower output wheel.

33. The method according to any one of claims 26, 27, 28 or 29 including the step of feeding the gas from said combustion chamber to a liquid seal blower output wheel.

34. The method according to any one of claims 26, 27, 28, or 29 including the step of feeding the gas from said combustion chamber to a dynamic turbine output wheel.

35. The method according to any one of claims 26, 27, 28 or 29 wherein said engine is a closed cycle engine including a closed conduit for continuously circulating a working gas in a closed cycle from said compressor to said output wheel and then back again to said compressor, and wherein said combustion chamber is an indirect combustion chamber, and including the step of indirectly heating said working gas in said combustion chamber.

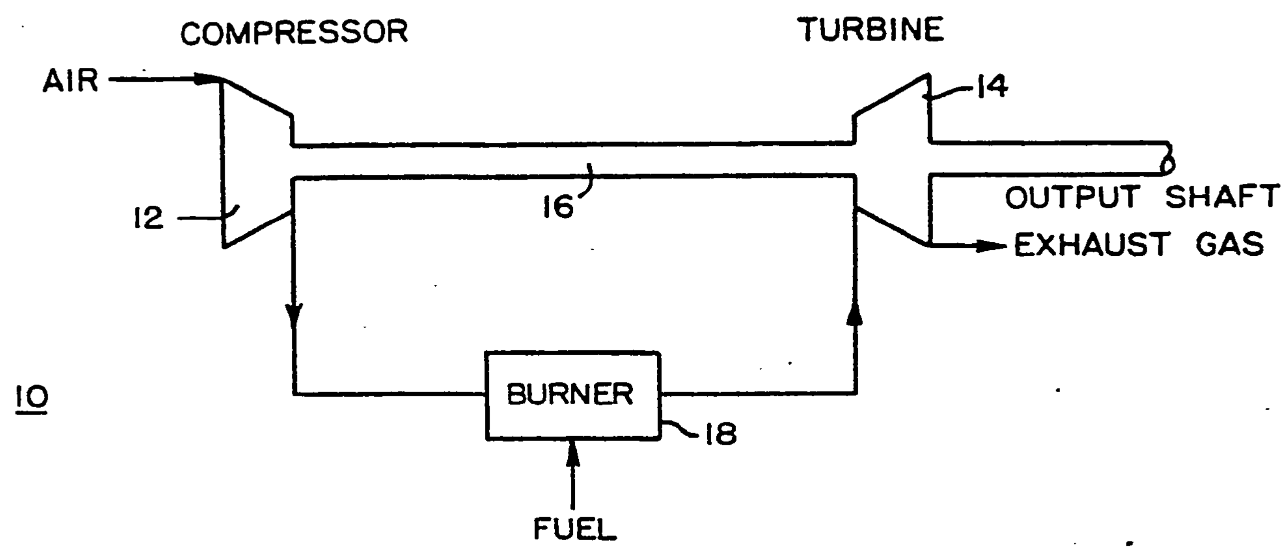

FIG. 1

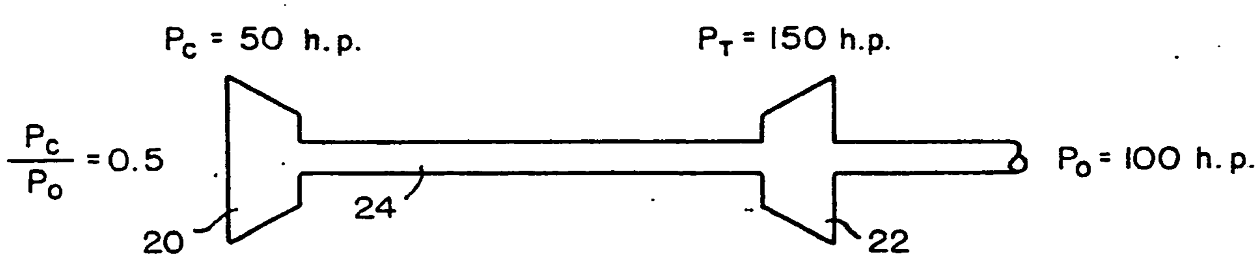

FIG. 2A

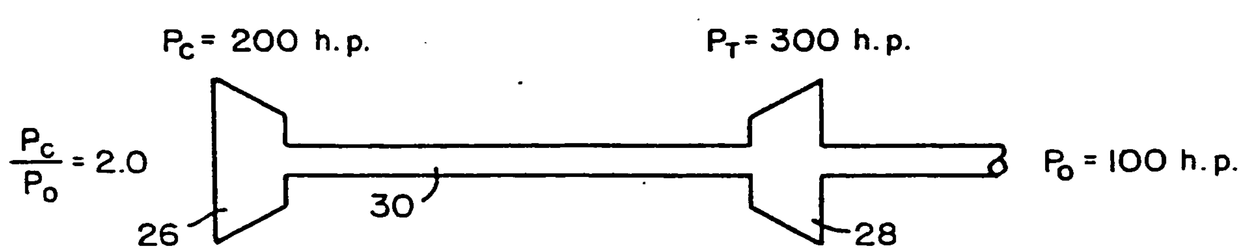

FIG. 2B

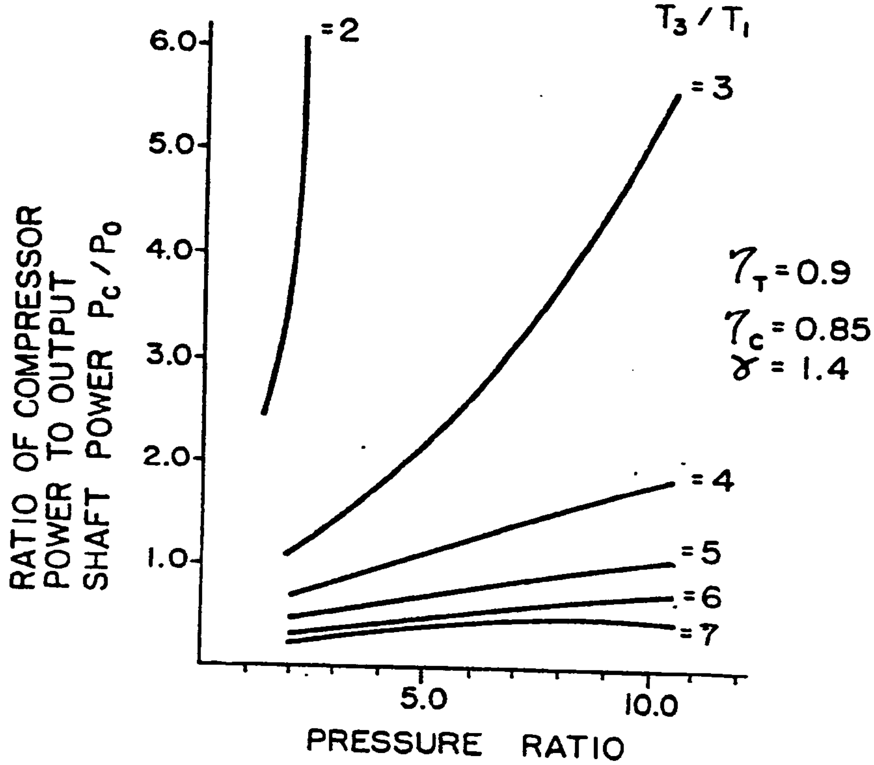
T3 / T1
=2
=3
=4
=5
=6
=7
$\eta_T = 0.9$
$\eta_c = 0.85$
$\gamma = 1.4$
6.0
5.0
4.0
3.0
2.0
1.0
RATIO OF COMPRESSOR POWER TO OUTPUT SHAFT POWER P_c / P_o
5.0
10.0
PRESSURE RATIO

FIG. 3

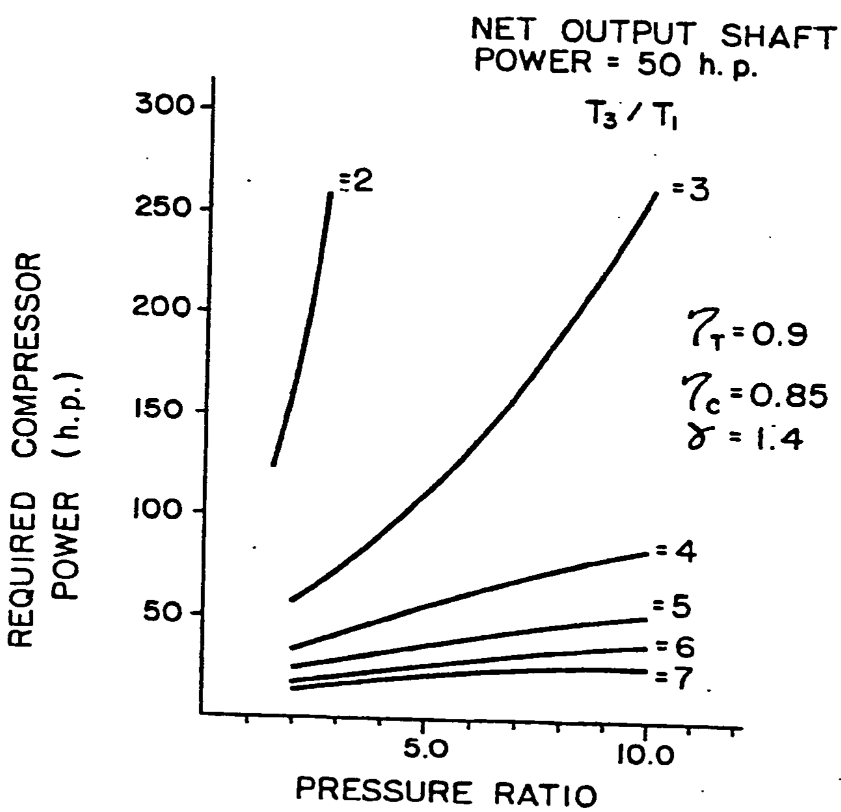
NET OUTPUT SHAFT POWER = 50 h.p.
T3 / T1
=2
=3
=4
=5
=6
=7
$\eta_T = 0.9$
$\eta_c = 0.85$
$\gamma = 1.4$
300
250
200
150
100
50
REQUIRED COMPRESSOR POWER (h.p.)
5.0
10.0
PRESSURE RATIO

FIG. 4

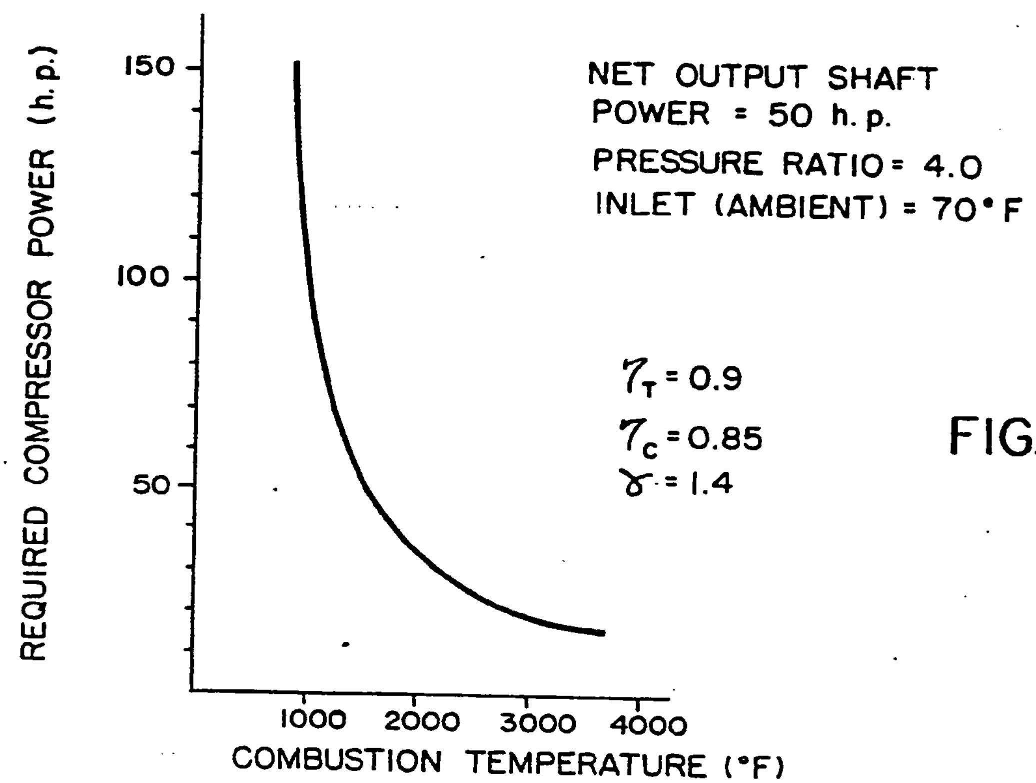
REQUIRED COMPRESSOR POWER (h. p.)
150
100
50
1000 2000 3000 4000
COMBUSTION TEMPERATURE (°F)
NET OUTPUT SHAFT POWER = 50 h. p.
PRESSURE RATIO = 4.0
INLET (AMBIENT) = 70° F
$\eta_T = 0.9$
$\eta_C = 0.85$
$\gamma = 1.4$

FIG. 5

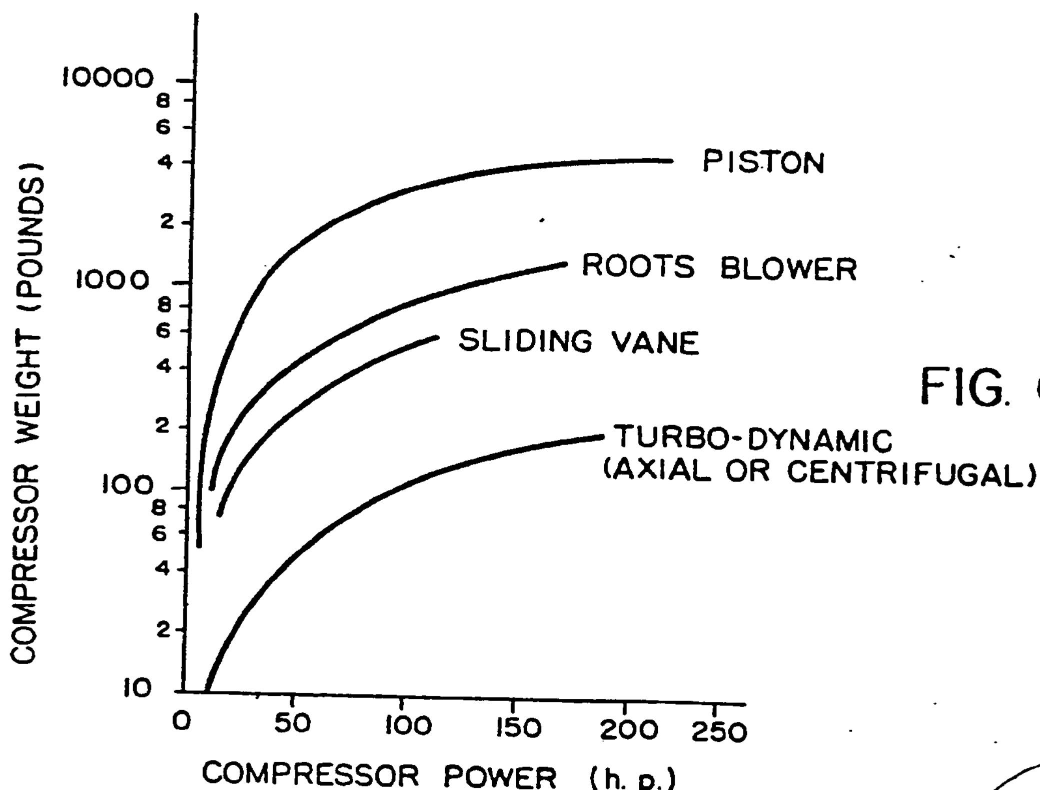
COMPRESSOR WEIGHT (POUNDS)
10000
8
6
4
2
1000
100
10
0 50 100 150 200 250
COMPRESSOR POWER (h. p.)
PISTON
ROOTS BLOWER
SLIDING VANE
TURBO-DYNAMIC
(AXIAL OR CENTRIFUGAL)

FIG. 6

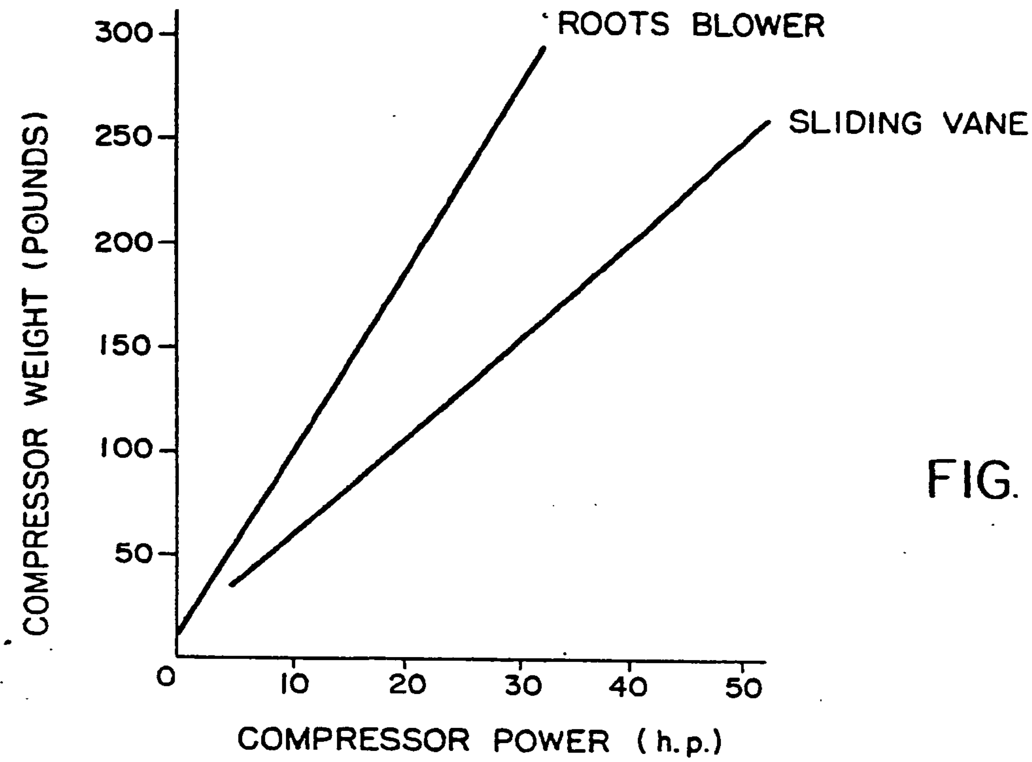
ROOTS BLOWER
SLIDING VANE
COMPRESSOR WEIGHT (POUNDS)
300
250
200
150
100
50
0
10
20
30
40
50
COMPRESSOR POWER (h.p.)

FIG. 7

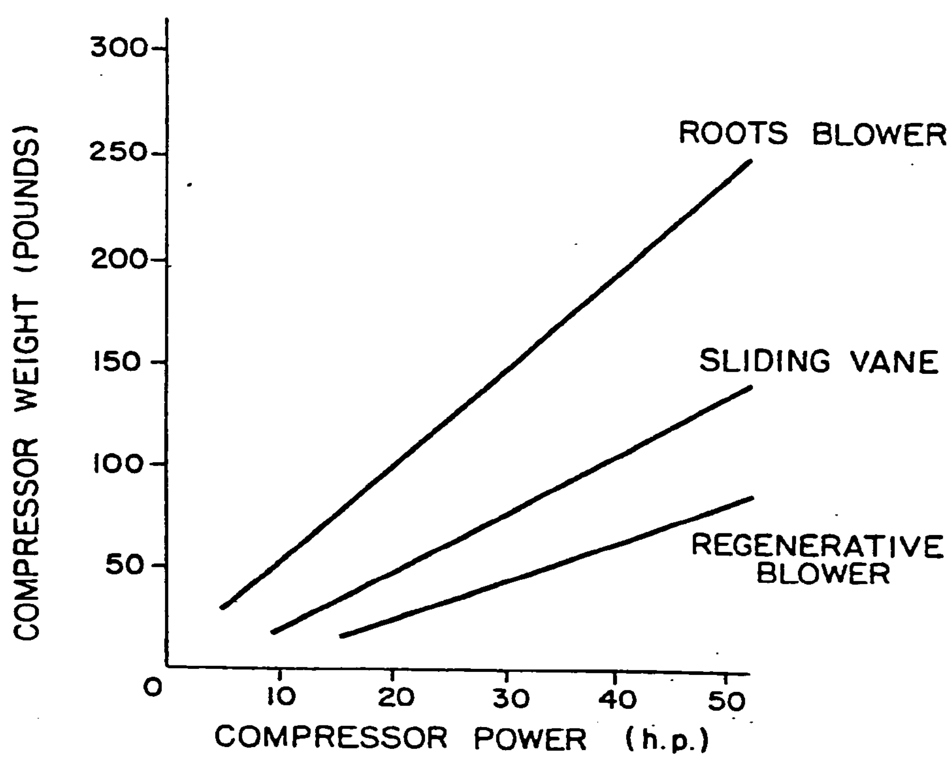
ROOTS BLOWER
SLIDING VANE
REGENERATIVE BLOWER
COMPRESSOR WEIGHT (POUNDS)
300
250
200
150
100
50
0
10
20
30
40
50
COMPRESSOR POWER (h.p.)

FIG. 8

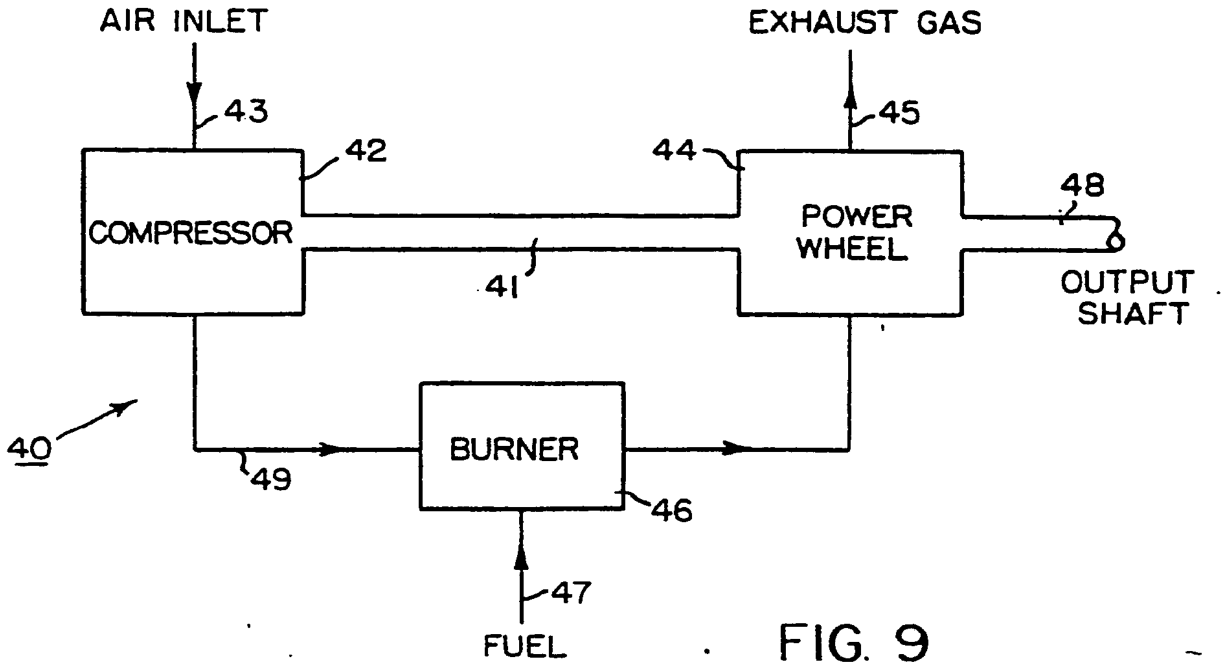
AIR INLET
43
42
EXHAUST GAS
45
44
COMPRESSOR
POWER WHEEL
48
41
OUTPUT SHAFT
40
BURNER
49
46
47
FUEL

FIG. 9

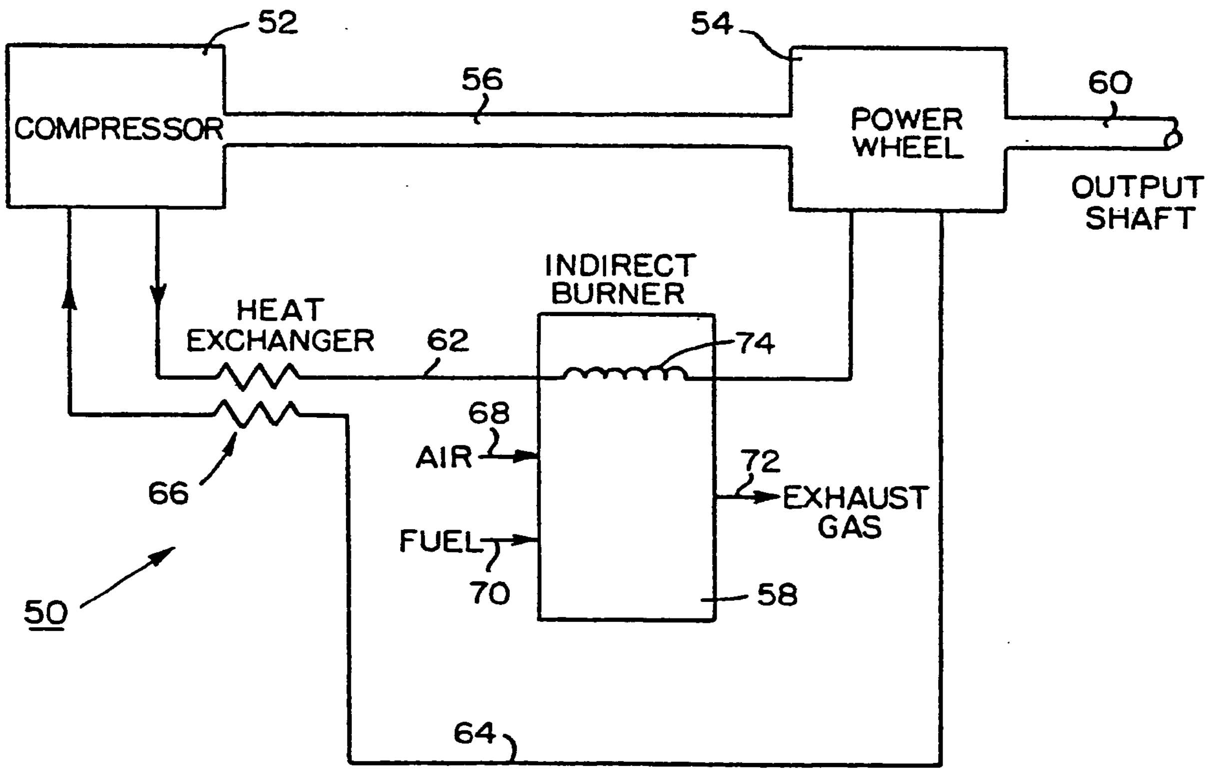
52
54
56
COMPRESSOR
POWER WHEEL
60
OUTPUT SHAFT
INDIRECT BURNER
HEAT EXCHANGER
62
74
68
AIR
72
EXHAUST GAS
66
FUEL
70
58
50
64

FIG. 10

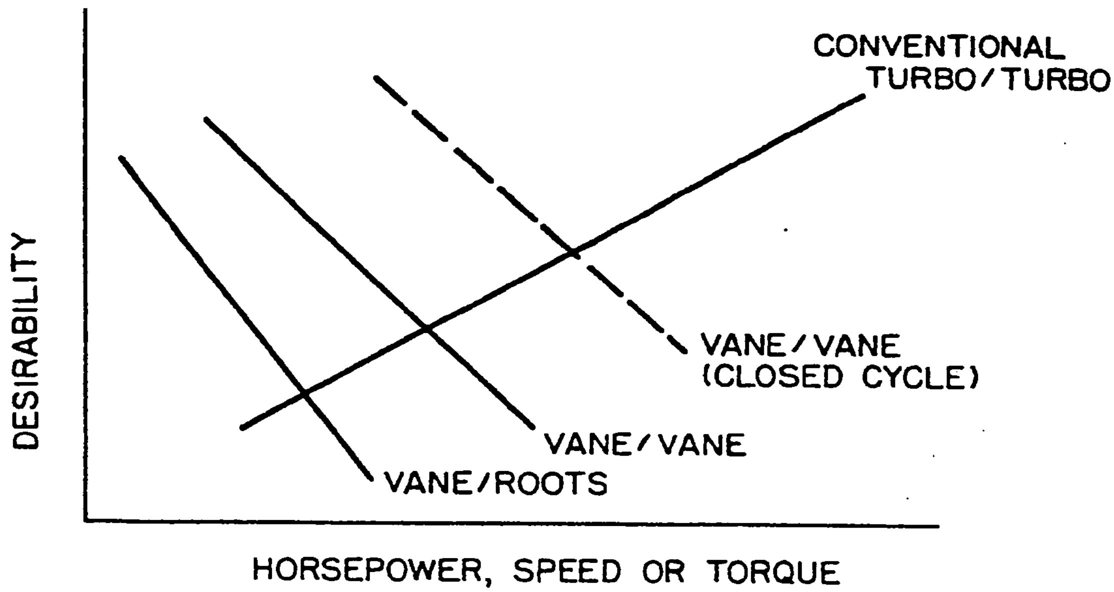
CONVENTIONAL
TURBO/TURBO
DESIRABILITY
VANE/VANE
(CLOSED CYCLE)
VANE/VANE
VANE/ROOTS
HORSEPOWER, SPEED OR TORQUE

FIG. 11

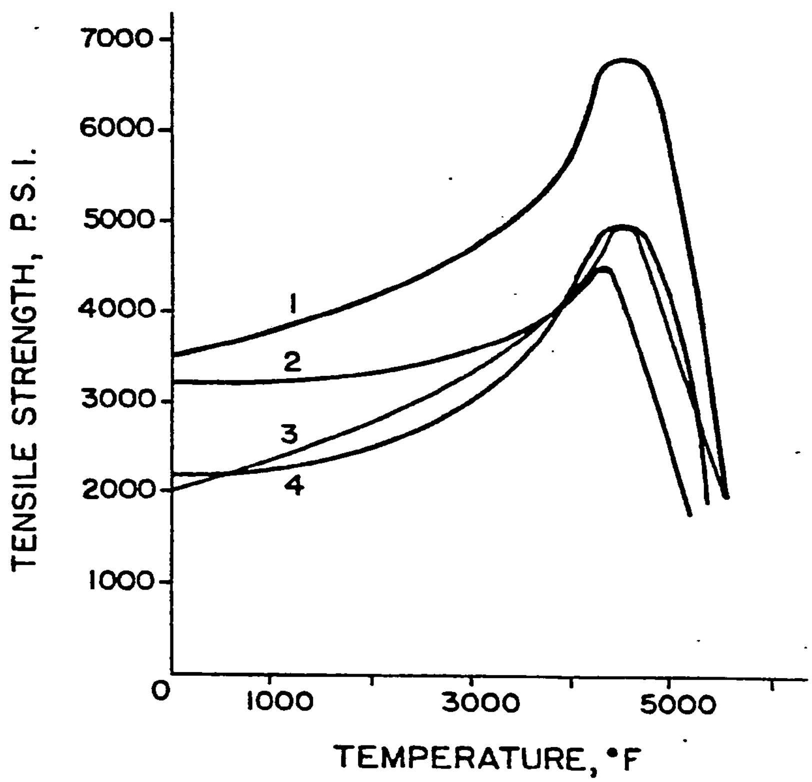
7000
6000
5000
4000
3000
2000
1000
TENSILE STRENGTH, P.S.I.
1
2
3
4
0
1000
3000
5000
TEMPERATURE, °F

FIG. 12

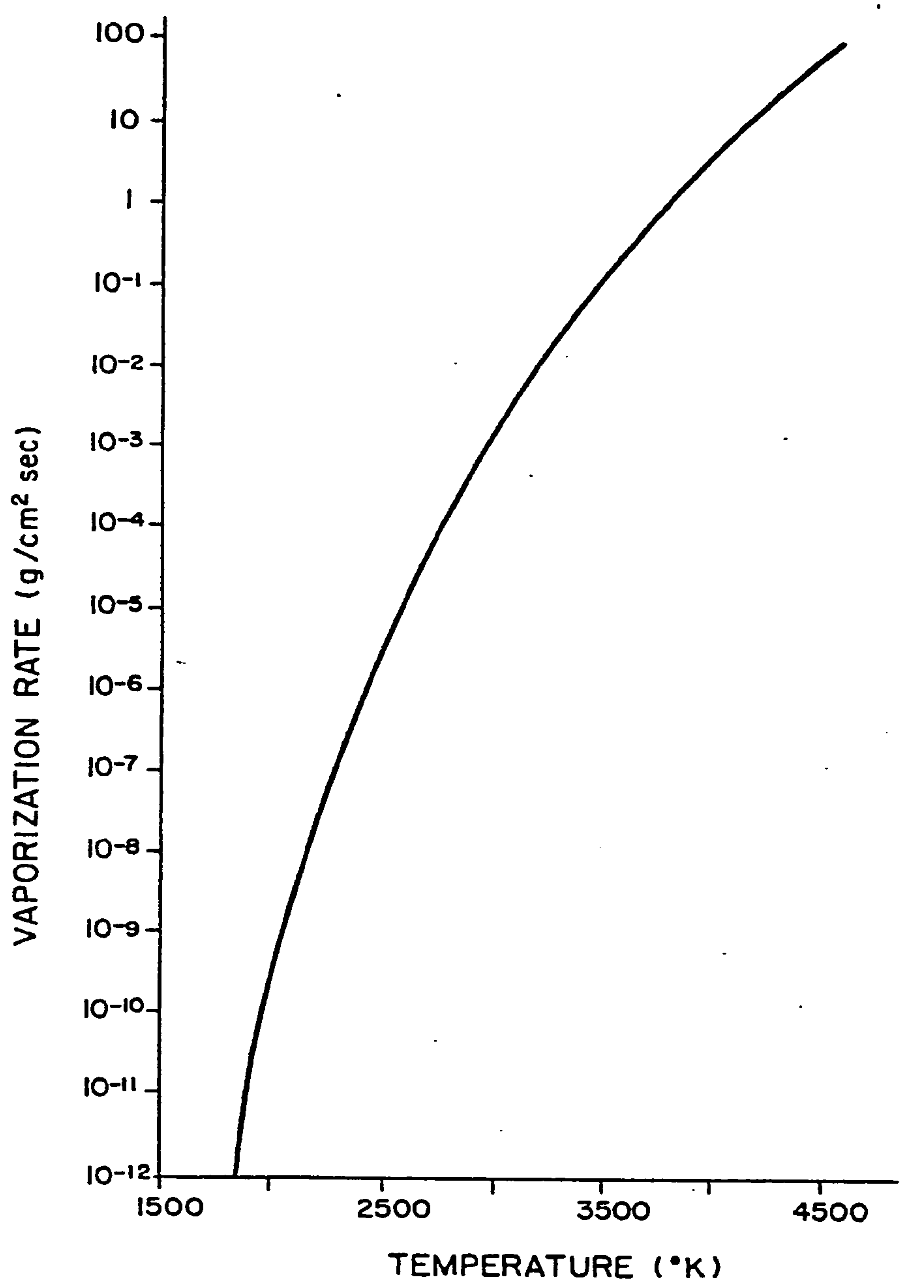
100
10
1
10^{-1}
10^{-2}
10^{-3}
10^{-4}
10^{-5}
10^{-6}
10^{-7}
10^{-8}
10^{-9}
10^{-10}
10^{-11}
10^{-12}
VAPORIZATION RATE (g/cm^2 sec)
1500
2500
3500
4500
TEMPERATURE (°K)

FIG. 13

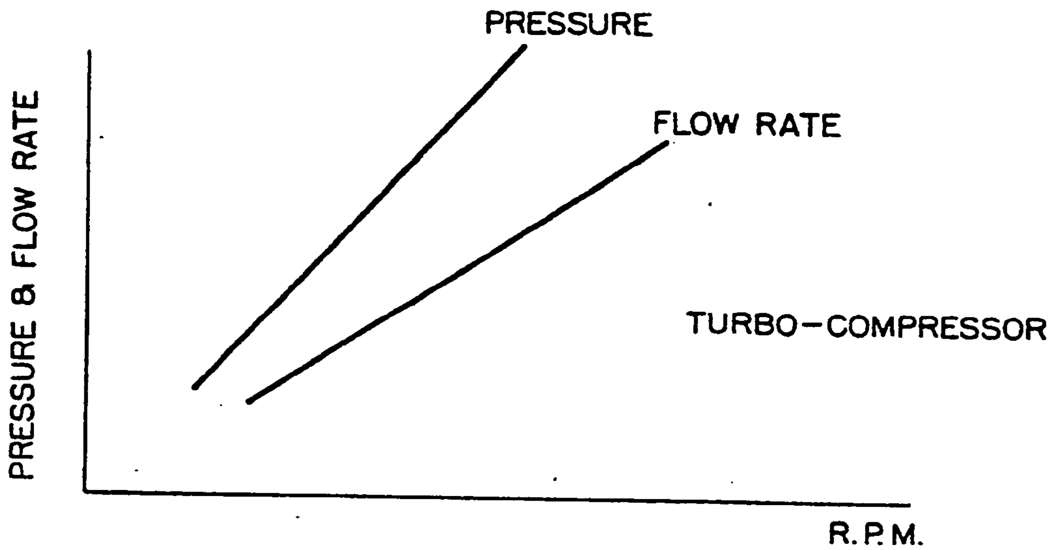
PRESSURE
FLOW RATE
PRESSURE & FLOW RATE
TURBO-COMPRESSOR
R.P.M.

FIG. 14

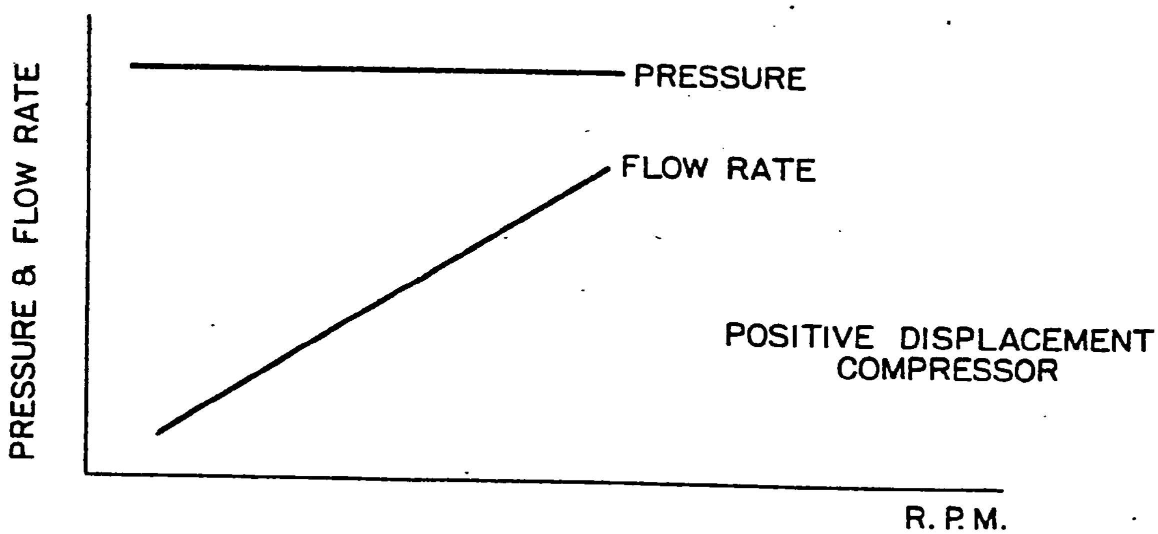
PRESSURE
FLOW RATE
PRESSURE & FLOW RATE
POSITIVE DISPLACEMENT
COMPRESSOR
R.P.M.

FIG. 15

INTERNATIONAL SEARCH REPORT WO 79/01071

International Application No PCT/US79/00324

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) [3]

According to International Patent Classification (IPC) or to both National Classification and IPC

Intl. Cl. F02C 3/04; F02C 5/02
U.S. Cl. 60/39.02; 60/39.45; 60/39.63; 60/39.76

II. FIELDS SEARCHED

Minimum Documentation Searched [4]

Classification System	Classification Symbols
U.S.	60/39.02, 39.45, 39.6, 39.63, 39.06, 39.76 123/202, 204

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched [5]

III. DOCUMENTS CONSIDERED TO BE RELEVANT [14]

Category *	Citation of Document, [16] with indication, where appropriate, of the relevant passages [17]	Relevant to Claim No. [18]
X	US,A, 3,584,459, Published 15 June 1971, Amann	1,2,4-13, 15,16,20-23, 26-32, 34
X	US,A, 3,057,157, Published 9 October 1962, Close	5,6,8-10, 12,26,27, 29-31,33
X	US,A, 3,191,852, Published 29 June 1965, Kaatz et al	15,16
X	US,A, 3,501,913, Published 24 March 1970, Brille	1,17-19
X	US,A, 3,672,164, Published 27 June 1972, Pieper	1,17-19
A	US,A, 2,461,757, Published 15 February 1949, Moores	1,2,4-13, 15-23,26-34
A	US,A, 3,765,171, Published 16 October 1973, Hagen et al	1,2,4-13, 15-23,26-34
A	US,A, 3,558,236, Published 26 January 1971, Bylsma	7,9-14,16, 28,30-35

* Special categories of cited documents: [15]

"A" document defining the general state of the art

"E" earlier document but published on or after the international filing date

"L" document cited for special reason other than those referred to in the other categories

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but on or after the priority date claimed

"T" later document published on or after the international filing date or priority date and not in conflict with the application, but cited to understand the principle or theory underlying the invention

"X" document of particular relevance

IV. CERTIFICATION

Date of the Actual Completion of the International Search [2]	Date of Mailing of this International Search Report [2]
17 August 1979	19 SEP 1979
International Searching Authority [1]	**Signature of Authorized Officer** [10]
ISA/US	Carlton C. Croyle

Form PCT/ISA/210 (second sheet) (October 1977)

International Application No. PCT/US79/00324

FURTHER INFORMATION CONTINUED FROM THE SECOND SHEET

A	US,A, 3,973,865, Published 10 August 1976, Mugele	7,9-14,16, 28,30-35
X	GB,A, 947,822, Published 29 January 1964, Birmann	1,17-19
A	AT,B, 140,501, Published 11 February 1935, Pistl	1,2,4-13, 15-23,26-34

V.☐ OBSERVATIONS WHERE CERTAIN CLAIMS WERE FOUND UNSEARCHABLE [10]

This international search report has not been established in respect of certain claims under Article 17(2) (a) for the following reasons:

1.☐ Claim numbers ______, because they relate to subject matter [12] not required to be searched by this Authority, namely:

2.☐ Claim numbers ______, because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out [13], specifically:

VI.☒ OBSERVATIONS WHERE UNITY OF INVENTION IS LACKING [11]

This International Searching Authority found multiple inventions in this international application as follows:

I. Brayton cycle engine operating in open cycle.
II. Brayton cycle engine operating in closed cycle.

1.☐ As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims of the international application.

2.☐ As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims of the international application for which fees were paid, specifically claims:

3.☒ No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claim numbers: 1,2,4-13,15-23,26-34

Remark on Protest

☐ The additional search fees were accompanied by applicant's protest.

☐ No protest accompanied the payment of additional search fees.